ASME HPHT Standards

BSEE Domestic and International Standards Conference: Offshore Standards Update

28 January, 2014

J Robert Sims

Becht Engineering Co., Inc.

bsims@becht.com

Background

- Most of the equipment used in oil and gas drilling and production does not fit the definition of a pressure vessel and is therefore not within the scope of the ASME Pressure Vessel Codes, including Division 3.

  - ✓ However, selected rules from pressure vessel codes, such as ASME Section VIII, Division 2 have traditionally been referenced in the API standards that cover this equipment.

  - ✓ The older (pre 2007) Division 2 rules used linear-elastic design methods, which may not be appropriate for HPHT equipment that may undergo significant plastic deformation in areas of local stress concentrations.

  - ✓ However, most of the rules for design in Division 3 are applicable to HPHT equipment.

  - ✓ The 2007 and later Editions of Division 2 may also be applicable, except that design margins are higher and fracture mechanics is not included.

Background and Purpose of Presentation

- The design/analysis rules in ASME Section VIII, Division 3, together with the design margins in that document, have been proposed for application to the analysis of HPHT equipment.

- This presentation provides a discussion of the Division 3 design/analysis rules with emphasis on applicability to HPHT equipment with design pressures of 15,000 psi and higher.

Scope of Division 3 Application

- ◆ Division 3 covers design, construction, inspection, and overpressure protection of pressure vessels with design pressures generally above 10 ksi (70 MPa).

    - ✓ However, Div. 3 can be used for any design pressure.

- ◆ Division 3 can be used for fixed vessels, "mobile vessels" or vessels in transport vehicles.

Scope of Division 3 Application (continued)

- Division 3 is referenced in B31.3, Chapter IX on *High Pressure Piping* for design/analysis of high pressure piping components, particularly for fatigue analysis.

- Division 3 is used worldwide for high pressure vessels, since there is no other comparable Code.

  - ✓ Other countries, particularly Japan, are developing high pressure vessel codes based on Div. 3. The rules are expected to be very similar.

Current Applications of Division 3

◆ Hot and cold isostatic pressing (>50 years experience)

  ✓ Up to about 30 inch ID and 30 feet or more long

  ✓ Design pressure in the range of 15-30 ksi.

  ✓ Design temperature in the range of ambient to 500°F.

  ✓ Threaded or clamp and hub end closures

  ✓ 10 or more cycles per day in many cases

◆ Food sterilization (>20 years experience)

  ✓ Up to about 15 inch ID and 12 feet or more long

  ✓ Design pressure in the range of 60-100 ksi.

  ✓ Design temperature in the range of ambient to 150°F.

  ✓ Externally supported plug type end closures

  ✓ 10 or more cycles per day in many cases

Current Applications of Division 3 (continued)

- ◆ Quartz crystal growing (>50 years experience)
  - ✓ Up to about 36 inch ID and 50 feet or more long
  - ✓ Design pressure in the range of 15-25 ksi.
  - ✓ Design temperature in the range of ambient to 750°F.
  - ✓ Threaded or clamp and hub end closures
  - ✓ 1 cycle every few weeks
- ◆ Polyethylene production (>65 years experience)
  - ✓ Vessels up to about 30 inch ID and 20 feet or more long
  - ✓ Square and rectangular piping elbows, tees, reducers, etc. (block fittings).
  - ✓ Piping and vessel flanges and other fittings.
  - ✓ Design pressure in the range of 15-55 ksi.
  - ✓ Design temperature in the range of ambient to 650°F.
  - ✓ High frequency (e.g. 5 Hz) cycles up to about 15% of design pressure due to pressure pulsations.

Current Applications of Division 3 (continued)

- Oil and gas production
  - ✓ Sampling vessels with frequent pressure cycles.
  - ✓ Design pressure in the range of 10-30 ksi
- Hydrogen transport and storage (primarily composite wrapped pressure vessels)
  - ✓ Design Pressure 10-15 ksi
- Research and Development (>80 years experience)
- Lower pressure applications include:
  - ✓ Hydroprocessing vessels for the refining industry (hydrofiners and hydrocrackers) (1-3 ksi)
  - ✓ Natural gas transport and storage (3-4 ksi)

Background

- A series of papers presented at ASME PVP conferences in the mid to late 1970s described several failures that had occurred in high pressure vessels.

- Several of the authors of those papers recommended that ASME produce codes and standards to cover high pressure vessels.

- ASME formed an ad hoc working group in 1979 to study the issue. Their recommendations are on the next slide.

Background (continued)

- Ad hoc working group recommendations were:
  - ✓ Develop a high pressure vessel code for application at pressures generally above about 10 ksi (70 MPa) where the existing Section VIII, Division 1 and Division 2 were not suitable.
    - ❖ Division 1 can be used for design pressures in excess of 10 ksi (70 MPa), but does not result in an optimum design.
    - ❖ Division 2 was similar to Division 1 in that regard prior to the major rewrite that was published in the 2007 Edition. However, it does not incorporate fracture mechanics analysis, which is important to high pressure vessel design.
  - ✓ Add a chapter to ASME B31.3 to cover high pressure piping. Chapter IX was published in 1985.
  - ✓ Develop a high pressure systems code to cover aspects other than piping and vessels.

Background (continued)

- The Special Working Group on High Pressure Vessels (SWG HPV) was established in late 1980, early 1981.
  - ✓ Initial focus was on forged, non-welded vessels for the high end of the pressure range.
    - ❖ An extensive body of technical literature was available because of work done to support the high pressure polyethylene production facilities started by ICI and BASF in the late 1930s and early 1940s.
  - ✓ Based on stakeholder input, the decision to incorporate welded construction and material product forms other than forgings was made several years after the SWG was formed.

Background (continued)

- After over 15 years of dedicated volunteer work, Section VIII, Division 3 was published in 1997.

- Significant updates and modifications have been made since that time.

- This presentation covers the current code rules as published in the 2013 Edition.

- The SWG is now the Subgroup on High Pressure Vessels (SG HPV) reporting to the Standards Committee on Pressure Vessels.

Advantages of Division 3 for Design of HPHT Equipment

- Division 3 has only a limited number of design rules. The intent is to use design-by-analysis in most cases.

- Elastic-plastic analysis using a true stress – true strain material curve is required for thick wall components and is strongly recommended for all components.

- The elastic plastic analysis rules consider two failure modes:

  - ✓ Global failure, where the component can no longer support the applied load and the deformation increases without bound.

  - ✓ Failure due to local strain limit damage, where the material begins to show internal damage in the form of microvoid formation or cracking due to internal strain in areas of local stress concentrations and high triaxial tensile stresses.

Advantages of Division 3 for Design of HPHT Equipment (cont.)

◆ Division 3 does not use allowable stresses per se, but places a design margin of 1.8 on the load that causes global or local failure in a FEA model. Both yield and tensile strength, as well as the stress-strain curve, are considered in the FEA.

◆ Charpy V-notch impact tests (toughness tests) are required in essentially all cases. For high pressure vessels, the failure mode of ductile burst is rarely encountered. It is more important to ensure that fast fracture does not occur, so the emphasis in Division 3 is on toughness rather than strength.

◆ Division 3 requires a fatigue analysis.

◆ The primary fatigue analysis method in Division 3 is fracture mechanics.

   ✓ It is assumed that a flaw equal in size to the largest flaw that would not be detected by the NDE method to be used exists in each component at the location of the greatest stress.

   ✓ The number of cycles required to grow the flaw from the assumed initial size to the critical size at which failure occurs is calculated

Organization of Presentation

- ◆ This presentation focuses on the design aspects of Div. 3 rather than fabrication, examination, testing, etc. because only the design rules have been proposed for application to HPHT equipment

  - ✓ Static pressure design using elastic-plastic analysis with local strain limits.

  - ✓ Linear-elastic analysis and limitations.

  - ✓ Fatigue analysis
    - ❖ S-N, Structural Stress and Fracture Mechanics

  - ✓ Comments on use of Tresca and von Mises yield criteria

  - ✓ Comments on Differential Pressure Design.

  - ✓ Summary and Conclusions

Elastic-Plastic Analysis Concepts

◆ FEA with large displacement theory (i.e. consideration of non-linear geometry) and the von Mises Yield function is used.

◆ FEA uses true stress-true strain curves that are based on the general type of material (e.g. ferritic, austenitic) and the minimum specified yield and tensile strength values.

✓ Equations for calculating the true stress-true strain curve are provided in Division 3. They are based on extensive experimental work.

Elastic-Plastic Analysis Concepts
(continued)

- The maximum expected weld offset (misalignment) and peaking should be modeled because that can have a significant effect on the strain limit damage (local acceptance criteria – see subsequent slides).

- A fine mesh should be used in areas of local stress concentration, such as described above.

  - ✓ As with all FEAs, a mesh refinement study should be done.

Elastic-Plastic Analysis Acceptance Criteria

- Global Criteria use a Load-Resistance Factor Design (LRFD) approach similar to Civil Engineering practice for design of structures.

  - ✓ Design margins are applied to various load combinations.

  - ✓ API are proposing to draft a table with modified load combinations to specifically cover the types of loads encountered in HPHT equipment

- Local Criteria are based on limiting the damage due to plastic strain at all points in the model to a level that is a function of the state of triaxial stress at that location.

  - ✓ This is based to some extent on testing of notched tensile specimens conducted by the PVRC. In a triaxial tension stress field, failure may occur at very low levels of plastic strain.

Comments on Local Elastic-Plastic Analysis Acceptance Criteria

◆ The local criteria are most likely to be violated in regions of notches, such as thread roots and thickness transitions with small radii.

✓ Details that have a high local stress concentration factor will probably also have a low strain limit.

❖ FEA models should have a fine mesh in these areas.

◆ The acceptable equivalent plastic strain (strain limit) will be increased significantly if a reduction of area is specified and entered.

✓ Many material specifications do not have a minimum specified reduction of area, so this must be specified by the purchaser separately.

Comments on Strain Limit Damage (SLD) Calculations

◆ The strain limit damage (SLD) is most conveniently calculated as a ratio of the total equivalent plastic strain from the FEA at a node to the maximum permitted strain at that same node.

◆ Some FEA programs (e.g. Abaqus) have the SLD calculation "built in" and can display the ratio in a contour plot.

✓ SLD can also be calculated by copying the FEA output for each node to a spreadsheet, but this can be cumbersome if there are a lot of nodes.

Elastic-Plastic Analysis Acceptance Criteria (continued)

◆ In addition to the global and local criteria, the analyst must consider serviceability criteria including, but not limited to, criteria that may be specified by the User. Examples include:

  ✓ Deformation at flanged joints that could result in leakage

  ✓ Deformations in a valve that could affect sealing.

  ✓ Deformations in a subsurface safety valve that can cause a failure to function properly.

◆ Hydrostatic test (or autofrettage) condition acceptance criteria should include the effect on serviceability.

Elastic-Plastic Analysis Acceptance Criteria (continued)

- ◆ A buckling analysis is required for components subjected to a compressive stress field.

- ◆ A ratcheting analysis is required.
  - ✓ Elastic – perfectly plastic material properties are used.
  - ✓ The loads are applied and removed in the proper sequence and one of the following criteria must be met:
    - ❖ No cyclic plasticity
    - ❖ Elastic core remains in the primary load bearing boundary
    - ❖ No permanent change in overall dimensions.

Linear-Elastic Analysis

- ◆ Linear-elastic analysis is performed in the same way as in Section VIII, Division 2, except that the basic design margin is 2/3 of yield at temperature.

  - ✓ Stress linearization and categorization are often problematic. Although much work has been published in this area, it requires a lot of experience and judgment to get it right.

  - ✓ Even if done properly, linear-elastic analysis gives only a rough approximation of the load capacity of a thick wall structure.

    - ❖ Division 3 does not permit linear-elastic analysis if the diameter ratio (D_o/D_i) is ≥ 1.25.

Fatigue Analysis

- Fatigue analysis in Division 3 can be done using the "traditional S-N method" or the new "structural stress method" (KD-3) only if leak-before-burst behavior can be demonstrated.

  - ✓ Otherwise the fracture mechanics method of KD-4 must be used.

  - ✓ The structural stress method is limited to the analysis of welds.

  - ✓ The "traditional S-N method" includes a mean stress correction approach.

 - ✓ Fracture mechanics should be used for all HPHT components.

Fatigue Analysis – Fracture Mechanics Method

◆ The fracture mechanics method (KD-4) is the most robust of the fatigue analysis methods provided in Division 3.

- ✓ Initial setup of the calculations can be complex, but after that it is relatively easy to use.

- ✓ The most accurate methods for calculating the crack tip stress intensity and reference stress for comparison to the failure assessment diagram (FAD) are found in API 579-1/ASME FFS-1, *Fitness-For-Service*.

- ✓ The range and mean of the stress intensity are used to calculate the crack growth rate using equations in Division 3.

- ✓ Fracture mechanics typically gives a longer life than either of the other methods for low cycle fatigue applications (e.g. less than 10,000 lifetime cycles).

25

Fatigue Analysis – Fracture Mechanics Method (continued)

◆ The assumption in the fracture mechanics method is that a flaw exists at the most highly stressed location in the structure immediately after construction.

  ✓ The depth and length of the initial flaw are based on the NDE method used.

    ❖ The largest flaw that could be undetected or would not be rejected is used as the starting point for the analysis.

    ❖ An initial flaw 1.6 mm ($^1/_{16}$ in) deep x 4.8 mm ($^3/_{16}$ in) long is typically used if WFMT or TOFD UT is used for detection, although these techniques can detect much smaller flaws if applied carefully.

    ❖ In the high cycle regime (e.g. >100,000 cycles), with relatively high cyclic stresses, assuming the above initial flaw can be very restrictive.

Fracture Mechanics Method – FAD (from API 579-1/ASME FFS-1)
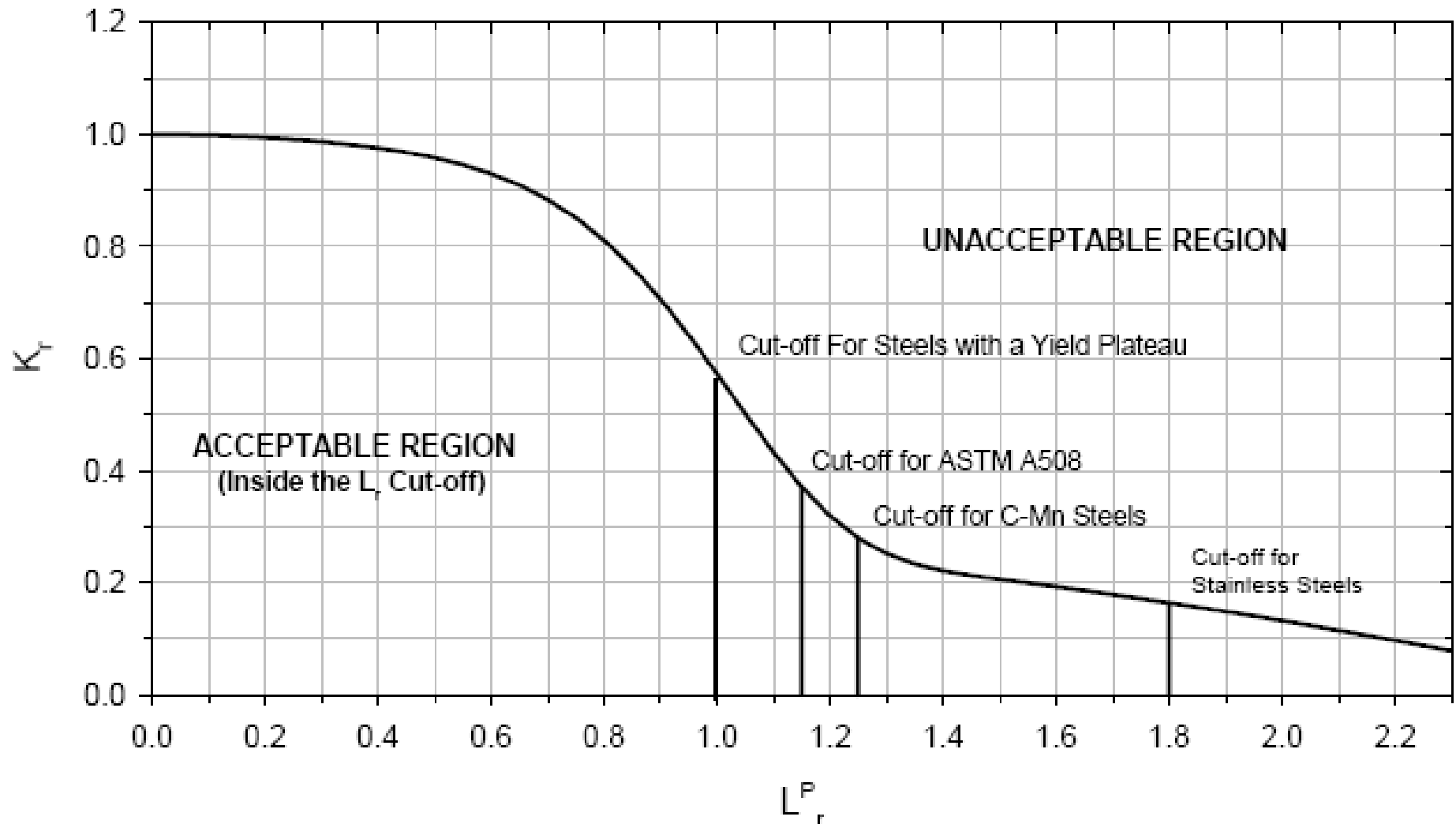
Fatigue Analysis – Fracture Mechanics Method (continued)

◆ The fracture mechanics method starts with a finite element analysis of the component with a fine mesh in the vicinity of local stress concentrations where cracks can be expected to initiate.

  ✓ The through thickness stress distribution from the FEA is copied to the fracture mechanics program.

  ✓ Weight function solutions as described in API 579-1/ASME FFS-1 are used to determine the range and mean of the crack tip stress intensity at the deepest point of the crack and at the component surface.

  ✓ The rate of crack growth per cycle (da/dN) is calculated.

Fatigue Analysis – Fracture Mechanics Method (continued)

- The number of cycles required to grow the crack by an amount that will result in an increase in crack tip stress intensity of less than about 1 to 2% is determined.

  - ✓ The crack growth calculation continues until the crack reaches a critical size (i.e. the boundary of the FAD).

- The stress intensity solutions are typically not valid for cracks deeper than 80% of the wall thickness, so if the crack reaches this depth before reaching the critical size, the analysis stops.

  - ✓ If this occurs, leak-before-burst (LBB) behavior can be assumed if the distance from the crack tip to the free surface (remaining ligament) is less than:
   - ❖ $(K_{Ic}/S_y)^2$ [from KD-141(a)(2)]

Fatigue Analysis – Fracture Mechanics Method (continued)

- The partial safety factors in API 579-1/ASME FFS-1 are not used in the analysis because the design margin is provided in Division 3 as the minimum of the following criteria:

  - ✓ ½ the number of cycles to the critical crack depth

  - ✓ Number of cycles to ¼ of the critical crack depth

  - ✓ Number of cycles to ¼ of the section thickness if LBB is demonstrated.

  - ✓ The later 2 criteria are very restrictive for thin wall sections and will probably be modified in the future.

Effect of Axial Stress on Through Thickness Yield Pressure

- In older methods of linear-elastic analysis, such as those in Section VIII, Division 2 prior to the 2007 Edition, linear elastic analysis used the stress difference (Tresca stress).

- More modern methods use the equivalent stress (von Mises stress)

- The next slide shows the effect of the intermediate principal stress, which is the axial stress in the case of a cylindrical component, on the plastic collapse pressure.

  - ✓ Within the range of axial stress shown, the collapse pressure varies by over 15%.

  - ✓ Obviously, axial stresses outside of that range will cause greater variations.

# Effect of Axial Stress on Through Thickness Yield Pressure
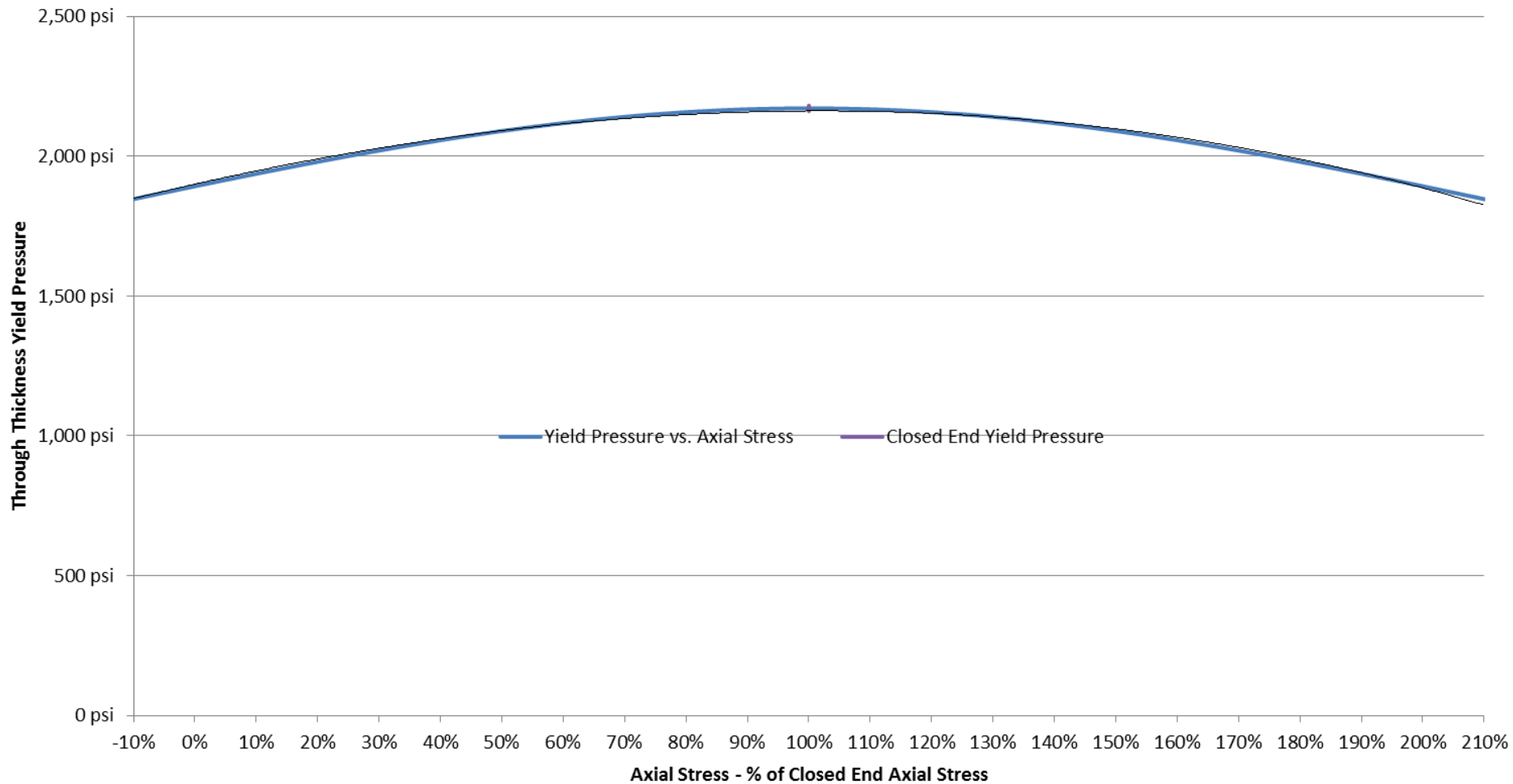
Differential Pressure Design

◆ When a component is subjected to both internal and external pressure, the design pressure is a function of the pressure difference.

◆ However, the design pressure cannot be calculated by simply using the pressure difference in design equations or in FEA.

 ✓ Actual values of internal and external pressure must be applied in an elastic-plastic FEA to determine the plastic collapse pressure in order to properly consider the effect of the increased radial compressive stresses on the failure pressure.

 ✓ Depending on the component, axial compressive stresses may also become significant.

Differential Pressure Design

- The next slide shows an example of the effect of external pressure.

- The top red line shows the internal pressure to cause through thickness yielding of a closed end cylinder if the closed form von Mises equation is used based on the difference between the internal and external pressures.

- The lower blue line shows an approximation, using a closed form solution, of the results of a finite element analysis that shows that the increased radial compressive stress that results from the combined internal and external pressure reduces the through thickness yield pressure, but not as much as ignoring the favorable effect of external pressure.

Effect of External Pressure on Through Thickness Yield Pressure
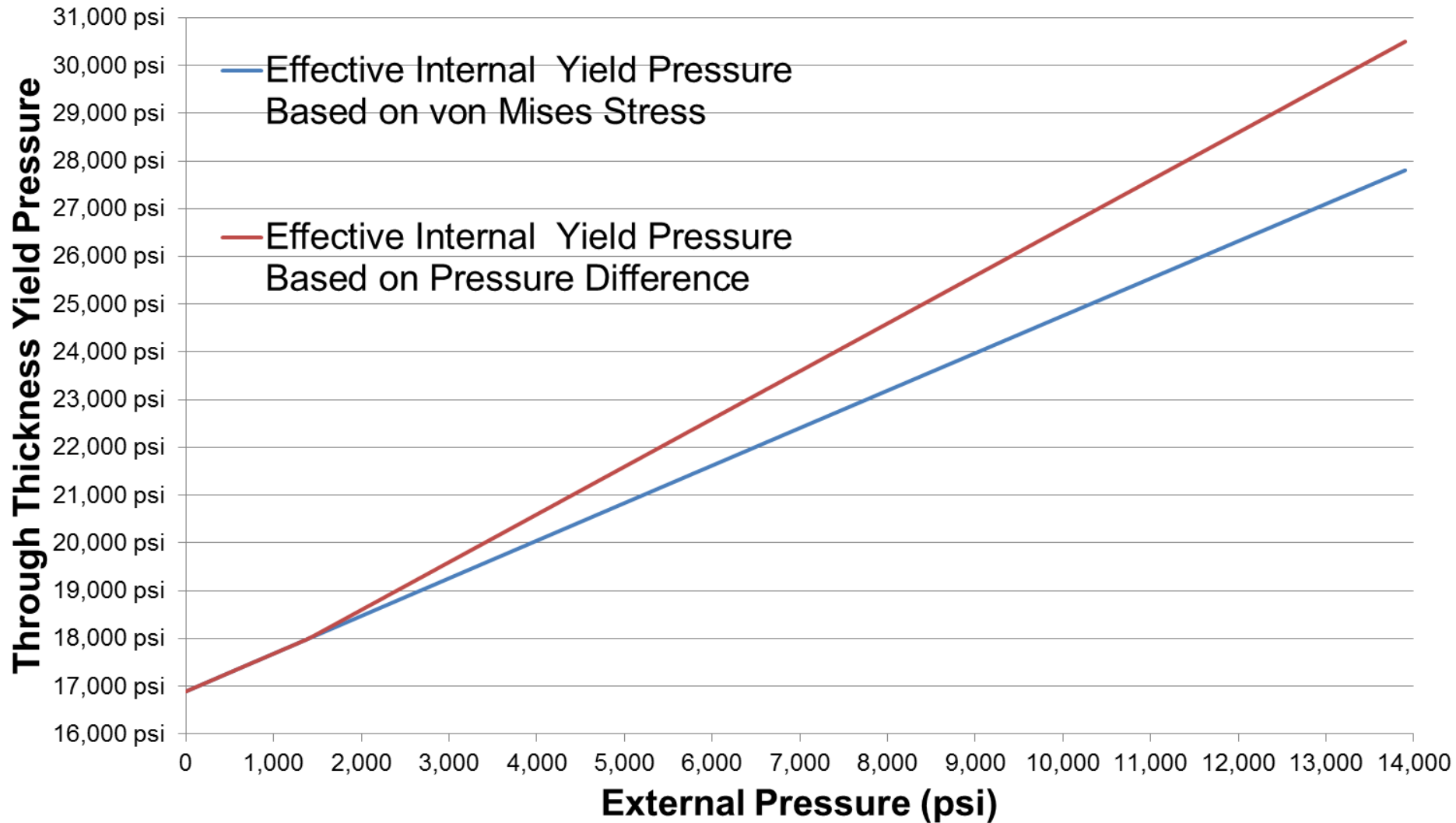
Other Comments on External Pressure

◆ The previous example showed a case where the external pressure only acted on the OD of a component. If the external pressure also produces axial compressive stresses, the effects will be different, but can still be determined accurately if FEA considering all loads is used.

  ✓ In addition, the effect of external pressure is different for open end (as compared to closed end) cylinders.

Summary and Conclusion

◆ Section VIII, Division 3 is a versatile Code that can be used in many applications. In some respects, it is an extension of the concept that led to the development of Section VIII, Division 2 in the late 1960's and the rewrite that was published in 2007.

◆ The design of HPHT equipment should be done to Division 3 to take advantage of the extensive experience and development work that went into the development of that Code.

◆ The effects of external pressure should be considered using elastic-plastic finite element analysis.